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The Effect of RME-1-Butanol Blends on Combustion, Performance and Emission of a Direct Injection Diesel Engine

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Abstract: The main objective of this study was assessment of the performance, emissions and combustion characteristics of a diesel engine using RME–1-butanol blends. In assessing the combustion process, great importance was placed on evaluating the stability of this process. Not only were the typical COV_{IMEP} indicators assessed, but also the non-burnability of the characteristic combustion stages: ignition delay, time of 50% heat release and the end of combustion. The evaluation of the combustion process based on the analysis of heat release. The tests carried out on a 1-cylinder diesel engine operating at a constant load. Research and evaluation of the combustion process of a mixture of RME and 1-butanol carried out for the entire range of shares of both fuels up to 90% of 1-butanol energetic fraction. The participation of butanol in combustion process with RME increased the in-cylinder peak pressure and the heat release rate. With the increase in the share of butanol there was noted a decrease in specific energy consumption and an increase in engine efficiency. The share of butanol improved the combustion stability. There was also an increase in NO_x emissions and decrease in CO and soot emissions. The engine can be power by blend up to 80% energy share of butanol.

Keywords: combustion stability; combustion stages; biodiesel; butanol; emission

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1. Introduction

Biodiesel as RME (rapeseed methyl ester) is a promising alternative fuels, which has the potential to reduce both the dependency on petroleum fuels and environmental pollution of using diesel fuel alone. Biodiesel is produced, inter alia, from food crops. This requires the use of fields for their production. On the other hand, alcoholic fuels can be produced from any waste biomass due to the production of alcohols being cheaper than producing biodiesel. Such fuels do not have a negative effect on the food market. The production of these fuels does not require extra land for cultivation. Such bioalcohols are considered as the next generation of alternative fuels [1]. For powering internal combustion engines both lower and higher alcohols are used. Lower alcohols, such as methanol or ethanol, create some inconvenience like low calorific values, phase separation and high value of latent heat of vaporization. Higher alcohols such as butanol or pentanol are closer to diesel fuel. These alcohols have no phase separation problems, higher calorific value—closer to fossil fuels and higher cetane number. However, comparing the properties of alcohols with those of diesel fuel, they cannot be used alone as a fuel to power a compression-ignition engine [2].

In the available literature, there can be found many research papers relating to the use of alcoholic fuels to power reciprocating internal combustion engines. Both for spark ignition engines, which can be powered by alcohol only, and compression ignition engines, which are fueled with dual fuel technology or with mixtures of fuels [3]. Alcoholic fuels have a very positive effect on the operation of compression ignition engines because they significantly reduce the emission of soot. This is due to the presence of oxygen in the alcohol particle structure, which promotes the oxidation of the soot particles. Here, as mentioned, dual-fuel technology can be used, but it involves some design changes in the engine's fuel system [4]. A simpler way is to feed the engine with a mixture of fuels [5]. Unfortunately, fuels such as ethanol or methanol do not mix with diesel fuel by itself. After a certain amount of alcoholic fuel (light alcohol) is exceeded, phase separation takes place over time. In this method of feeding, a better way is to select biofuels that have a similar molecular structure and thus have a greater mixability. There are many different types of biodiesels in use that can be mixed with alcohols without much difficulty. Biodiesel fuels can be of both plant and animal origin [6]. An interesting solution is the use of biodiesel and one of the so-called higher alcohols such as butanol to supply the engine.

The share of higher alcohol in fuel blend causes its properties. Butanol is considered as a promising fuel in diesel engine owing to its viscosity, higher calorific value compared to lower alcohols and it forms stable mixtures [7,8]. Goga et al. [8] presented investigation results of diesel engine powered by biodiesel fuel produced from rice bran and n-butanol. They studied engine operating parameters and emission. They stated that biodiesel and higher alcohol could help in improving the engine performance and decreasing harmful exhaust gases emission. They also stated that diesel engine without any change in the supply system can be powered by mixtures of biodiesel produced from rice bran and n-butanol. The specific fuel consumption increased, but in the case of combustion of different fuels, it is not a parameter that gives precise information. Another source of biodiesel co-combusting with butanol was tested by Killol et.al [9], Karanja methyl ester blended in small proportions with n-butanol up to 20%. Based on the obtained results, it was found, inter alia, that the fraction of n-butanol increases the ignition delay period up to 4 degrees. At full load of the engine, the time of the combustion duration decreases with an increase in the proportion of butanol in the mixture. Regarding exhaust emissions, the CO was decreased with the increase in butanol fraction. With the increase in the proportion of butanol in the mixture, NO_x emission first increased and for higher butanol proportions decreased [9]. Singh et al. [10] investigated the impact of n-butanol as an additive with eucalyptus biodiesel–diesel blends on the performance and emission characteristics of the diesel engine. They noticed near to 20% reduction in CO and 40% in HC emissions for B20 (20% of biodiesel—80% of diesel) compared to net diesel. With addition of butanol, reduction in NO_x emission was obtained but emission of CO_2 was higher than for diesel combustion. Jeevahan et al. [1] investigated the impact of 1-Butanol participation in blend on the engine performance and emissions characteristics. They stated that 1-butanol in the blend slightly reduced the brake thermal efficiency as compared to biodiesel combustion, but it causes a decline in specific fuel consumption and emissions of NO_x, CO and THC. As a conclusion, they stated that adding butanol to biodiesel is of benefit to a compression-ignition engine. [1]. Higher alcohols, such as butanol, are also mixed with diesel fuel. Atmanli and Yilmaz [11] investigated impact of butanol/diesel fuel blend on engine parameters. They stated that participation of 1-butanol (up to 35% v/v) causes an increase in brake specific fuel consumption by 14% and a decrease in brake thermal efficiency by 7% compared to an engine powered by diesel fuel. Rapeseed oil is often used to power compression ignition piston engines. Rman et al. [12] investigated rapeseed oil and diesel-biodiesel blend as a fuel for power diesel engine. Thy stated that the thermal efficiency of the engine powered by biodiesel and its blends was lower than for diesel fuel. The in-cylinder peak pressure and heat release rate of biodiesel blended fuels are lower in comparison with results obtained for the diesel fuel. In case of emission assessment, the share of biodiesel caused a decrease in THC and CO emission simultaneously with an increase in NO_x and smoke. A similar study conducted Qi et al. [13] stated that in the case of a low-load engine, the combustion process, for both the engine fueled with diesel fuel and fueled with a mixture with butanol, started in the same crank angle. For combustion with butanol, higher values of maximum pressure and HRR were recorded. In case of high load, similar values of maximum pressure and HRR were obtained for both fuels. The ignition delay time was shortened for the engine fueled with a blend. For low engine loads, the specific fuel consumption and the specific energy consumption were slightly higher. At high engine load, the soot emission was lower than for the engine powered by diesel fuel, and at low loads, it was slightly higher. Regarding the NO_x emission, there were no significant differences at high load, while at low engine load the NO_x emission was slightly lower. [13]. Nuortila et al. [14] investigated properties of 1-butanol blended with rapeseed oil up to 30% v/v. The blends were stable and have been assessed as fuels that meet the requirements for engine fuels. The vast majority of the analyzed research reports show that the butanol content is tested up to about 30% in volume [10,11,15,16].

The results included in the literature review do not directly relate to the assessment of the impact of the fuel type on the stability of the combustion process. The repetitive combustion process in the subsequent cycles of the engine's operation translates into stable operation of the engine and the device driven by the engine. Changing the engine fuel, and each fuel mixture is a different fuel, significantly affects the combustion stability. The paper presents results of experimental investigations of a diesel engine fueled by RME-1-butanol blends. The energy share of 1-butanol in the range from 0 to 90% was analyzed. The effect of such blends on the diesel engine performance and exhaust gas emission as well as combustion stability were studied. The combustion stability analysis was based on the COV_{IMEP} and the spread of characteristic stages of the combustion process.

2. Materials and Methods

The research carried out with the use of a compression ignition reciprocating internal combustion engine. It is one cylinder, four-stroke, air cooled, direct injection (DI) diesel engine with power of 5 kW at rated speed of 1500 rpm. The main specifications and diagram of the engine test stand are presented in Table 1 and Figure 1, respectively. The engine was coupled with a dynamometer to provide brake load. The already existing factory injection system was used to supply the engine with fuel. The test was carried out with a constant fuel injection angle and constant engine load. For each measurement point, a minimum of 3 times 200 consecutive engine work cycles were recorded. Average values from 200 cycles were used to calculate the engine operation indicators. The fuel consumption was controlled by measuring the consumption time of a known volume of fuel. Air consumption was also measured to determine the excess air ratio. The used measuring system allows on-line visualization of the pressure course, pressure increase, heat release, power, rotational speed and value of indicated mean effective pressure for the current engine cycle.

Table 1. Engine main specification.

Title 1	Title 2
Type of engine	4-stroke compression ignition
Number of cylinders	1
Bore	90 mm
Stroke	90 mm
Displacement volume	573 cm^3
Number of valves	2
Compression ratio	17
Engine speed	1500 rpm
Fuel injection	mechanical direct injection
Fuel injection timing	20 degress bTDC

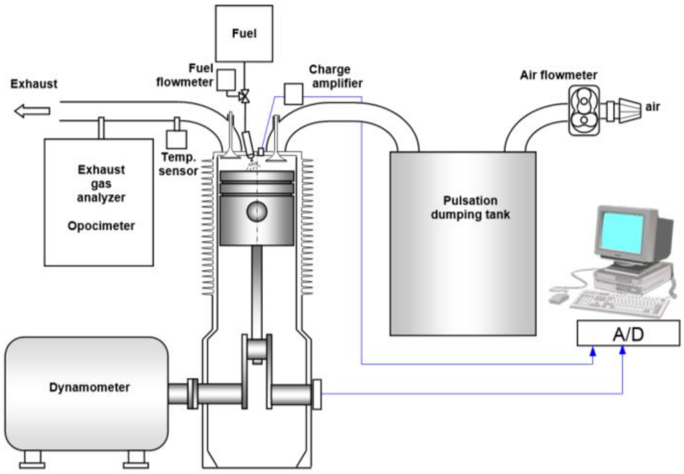


Figure 1. Diagram of the test stand.

Figure 1 shows a diagram of the test stand. The test engine is connected with a dynamometer ensuring maintenance of the assumed engine load. The engine's intake system is equipped with an air flow meter: air rotor flowmeter Common CGR-01 G40 DN50 (measuring range 0.65 . . . 65 m^3/h, accuracy class 1) and a pulsation dumping tank between the engine and the flow meter to suppress pressure pulsations. The in-cylinder pressure measurement path consists of: pressure sensor Kistler 6061, range 0 . . . 250 bar, linearity <± 0.5% FS and charge amplifier Kistler 511, range ± 10 ... ± 999,000 pC for 10V FS, error <± 3%, linearity <±0.05% FS. The signal from the measuring path is transferred to data acquisition module, Measurement Computing USB-1608HS—16 bits' resolution, sampling frequency 20 kHz with software. Exhaust emissions were controlled with Bosch BEA 350 analyzer and soot emission by smoke meter 415SE from AVL with measuring range to 10 FSN (filter smoke number), detection threshold 0.002 FSN or 0.02 mg/m^3 and repeatability ≤± 0.005 FSN plus 3%. The exact measurement data of the exhaust gas analyzer are given in Table 2.

Table 2. Parameters of the Bosch BEA 350 analyzer.

Apparatus	Measuring Range	Resolution	Accuracy from Measured Value	Absolute Accuracy
CO	0.000–10.000% vol.	0.001% vol.	. . .	. . .
	0.000–5.000% vol.	0.001% vol	±5%	±0.06% vol.
HC	0–9999 ppm vol.	1 ppm vol.	. . .	. . .
	0–2000 ppm vol.	1 ppm vol.	±5%	±12 ppm vol.
CO_2	0.00–18.00% vol.	0.01% vol.	. . .	. . .
	0.00–16.00% vol.	0.01% vol.	±5%	±0.5% vol.
O_2	0.00–22.00% vol.	0.01% vol.	. . .	. . .
	0.00–21.00% vol.	0.01% vol.	±4%	±0.1% vol.
NO	0–5000 ppm vol.	1 ppm vol.	±4%	±25 ppm vol.
	0–4000 ppm vol.	1 ppm vol.	±8%	±50 ppm vol.
λ	0.500–9.999	0.001	. . .	. . .
	0.700–1.300	0.001	±4%	. . .

2.1. Fuel Characteristics

Two fuels with different physicochemical properties were used in the research. RME biodiesel was used as the basic and reference fuel. RME is a fully renewable product that can be blended with fossil diesel [4,14]. 1-butanol was used as an alternative fuel. Butanol is an environmentally friendly, renewable fuel. This fuel is considered to be one of the most efficient fuels for an internal combustion engine, which enables the elimination of petroleum-derived fuel [17,18].

The properties of the fuels are presented in Table 3. Both fuels belong to the group of oxygenated fuels. One of the most important features of biofuels is the oxygen content in their molecular structure, which significantly affects the combustion process. Both fuels also have a similar calorific value. This is an advantage because in the injection control process, the injector opening time does not need to be extended too much to deliver a dose of fuel of the same energy with a high proportion of 1-butanol [19,20]. Butanol has 2 times higher value of the heat of vaporization in relation to RME and the fact that both fuels have similar energy, the volume dose will not increase significantly and the cooling effect will not be significant. Fuel properties of 1-butanol (LHV:33.2 MJ/kg) are close to RME (LHV:37.1 MJ/kg); however, this does not allow the 1-butanol to be used directly in diesel engines. However, in the form of a mixture of both fuels, such a possibility exists. In the conducted tests, the energy content of 1-butanol was increased every 10% up to the flammability limit. Table 4 shows the fuel properties of RME-1-butanol blends. With an increase in the proportion of 1-butanol fraction, the oxygen fraction, cetane number, kinematic viscosity, LHV and density of the blends decreased. Among mentioned parameters, viscosity and density values are within the limits (EN590 standard) for fuels suitable for diesel engine. RME or blends of RME and alcohols, can be used for fueling existing diesel engines without chemical conversion [14].

Table 3. Fuel specifications.

Parameter	RME	1-Butanol
Chemical formula	$CH_3(CH_2)_nCOOH_3$	$C_4H_{10}O$
Cetane number	56	17–25
Density at 1 atm and 15 °C (kg/m^3)	855	810
Lower heating value (MJ/kg)	37.1	33.2
Heat of evaporation (kJ/kg)	250	585
Auto-ignition temperature (°C)	>101	343
Flash point (°C)	91–135	35
Stoichiometric air-fuel ratio	12.5	11.2
Kinematic viscosity at 40 °C (mm^2/s)	4.51	2.63
Oxygen content (wt%)	10.8	21.6

The following nomenclature has been used: B100—RME, BB10—10% of 1-butanol and 90% of RME in energy fraction, successive mixtures, respectively. With increase in the fraction of 1-butanol in the mixture, the calorific value decreased, which required the supply of a higher volume dose of fuel ensuring the same energy dose for the engine.

Figure 2 presents the energy doses supplied to the engine. It is visible that with the increase in 1-butanol fraction in the blend with RME the energy demand of the engine decreased. For BB80, which was the last proportion of butanol acceptable for the engine, the energy dose was 923.5 J and it was lower by 147.7 J compared to RME.

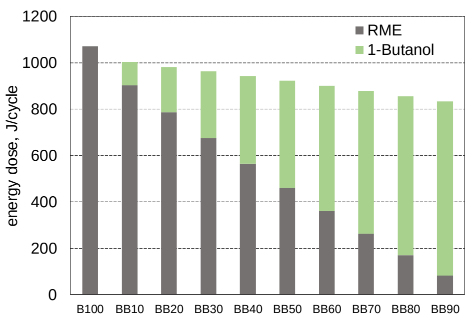


Figure 2. Composition of the fuel energy dose.

Table 4. Composition of fuel blends.

Properties	B100	BB10	BB20	BB30	BB40	BB50	BB60	BB70	BB80	BB90
RME, (% energy)	100	90	80	70	60	50	40	30	20	10
1-Butanol, (% energy)	0	10	20	30	40	50	60	70	80	90
LHV, (MJ/kg)	37.1	36.7	36.3	35.9	35.5	35.1	34.7	34.4	34.0	33.6
Heat of evap., (kJ/kg)	250	283.5	317	350.5	384	417.5	451	484.5	518	551.5
Oxygen content, (%)	10.8	11.88	12.96	14.04	15.12	16.2	17.28	18.36	19.44	20.52

2.2. Calculation Methodology

In the study, the indication results and the measured concentrations of toxic exhaust gas components as well as opacity were used to characterize the engine powered by rapeseed oil–butanol blends. The test results were obtained at the same engine operating conditions; therefore, the obtained results are comparable. The tests were carried out for a constant, unchanging engine load. The basic parameters of the engine's work, such as efficiency, indicated mean effective pressure, specific energy consumption, were determined. The LHV for the test fuels were different due to difference in chemical composition. The influence of butanol content in the mixture with RME was investigated for the proportions from 10 to 90% of butanol in energy share.

The mean effective pressure was determined based:

$$\mathrm{IMEP} = \frac{1}{\mathrm{V_d}} \int_0^{720} \mathrm{p} \frac{\mathrm{dV}}{\mathrm{d}\varphi}, \tag{1}$$

where: V_d—displacement volume [m^3].

The indicated thermal efficiency is defined as the ratio of the indicated work in the cylinder volume, averaged during the measurement of fuel consumption to the average amount of heat supplied to the cylinder.

$$\mathrm{ITE} = \frac{\mathrm{IMEP} \cdot \mathrm{V_d}}{\mathrm{Q_e}} 100\%, \tag{2}$$

where: IMEP—indicated mean effective pressure [Pa] and Q_e—total heat supplied to the engine in fuel dose per cycle [J].

One of the most important aspects of the evaluation of the combustion process in an engine is the analysis of heat release. Heat release rate ($dQ/d\varphi$) is determined on the basis of the in-cylinder pressure. The heat release rate was calculated on the basis of the first law of thermodynamics and the equation of state. After some simplifications, the net heat release is calculated based on:

$$\frac{\mathrm{dQ}}{\mathrm{d}\varphi} = \frac{1}{\kappa - 1}\left[\kappa \mathrm{p} \frac{\mathrm{dV}}{\mathrm{d}\varphi} + \mathrm{V} \frac{\mathrm{dp}}{\mathrm{d}\varphi}\right], \tag{3}$$

where: κ—the ratio of specific heats, V—cylinder volume and p—in cylinder pressure.

Usually, the uniqueness of the engine operation is determined by the COV_{IMEP} index, which describes the cycle-by-cycle variations of combustion process. The COV_{IMEP} is determined on the basis of IMEP data from several dozen consecutive engine work cycles. In this study the COV_{IMEP} is determined on the basis of 200 cycles. The COV_{IMEP} is directly related to the combustion stability. The COV_{IMEP} is defined as:

$$\mathrm{COV_{IMEP}} = \frac{\sigma_{\mathrm{IMEP}}}{(\mathrm{IMEP})_{\mathrm{mean}}} \cdot 100\%, \tag{4}$$

where: σ_{IMEP}—standard deviation of IMEP.

According to the literature, for correct operation of IC engine, the COV_{IMEP} should be lower than 5% [21].

The probability density distribution function is also used to assess the combustion process stability. This function as a probability density of IMEP is determined based on:

$$f(\mathrm{IMEP}) = \frac{1}{\sigma_{\mathrm{IMEP}}\sqrt{2\pi}} \exp\left(\frac{-\left(\mathrm{IMEP}_i - \overline{\mathrm{IMEP}}\right)^2}{2\sigma^2_{\mathrm{IMEP}}}\right), \tag{5}$$

To evaluate the combustion process, the ignition delay time, duration of combustion and, very often, the angle of occurrence of 50% of the heat released after TDC, are determined. In a compression-ignition engine, the ignition delay period is defined as the time from the start of fuel injection until the release of 10% of heat. The duration of combustion is defined as the time from release of 10% to 90% of heat release. To determine these characteristic stages of combustion, the average course of several dozen consecutive engine operation cycles is usually used. A reciprocating engine is a thermal machine with many factors contributing to its unstable operation. Combustion instability is classified into three categories:

- chamber instabilities—due to the occurrence of combustion inside a chamber (shock instabilities, fluid-dynamic instabilities associated with the chamber);
- intrinsic instabilities—chemical-kinetic instabilities, diffusive-thermal instabilities, hydrodynamics instabilities;
- system instabilities—caused by feed-system interactions, exhaust/intake-system interactions.

In the combustion chamber of IC engines all three types occur simultaneously. Phenomena affecting combustion instability in the engine cylinder: turbulence, vortex, fuel injection, movement of piston, movement of valves, exhaust gas recirculation, cylinder filling, etc.

In order to properly control the engine and obtain its repeatable operation in subsequent cycles, it is necessary to ensure combustion stability. Therefore, it is important to study this stability.

Cyclic combustion variability has nonlinear deterministic structure depending on engine operating conditions and combustion modes. Combustion instability is caused by physical phenomena occurring in a reacting flow. For these reasons, the specific characteristic combustion stages are only representative values, as they are burdened with the uncertainty error of their determination. Determining these ambiguity intervals gives real knowledge of possible changes in the combustion stages.

3. Results

The evaluation of the combustion process in an engine fueled with RME mixed with 1-butanol was based mainly on the analysis of the heat release. The nature of the heat release gives a lot of valuable information about the combustion process. The analysis of the uniqueness of the combustion process in the subsequent cycles of the engine's operation was also taken into account to a large extent. The assessment of combustion is usually performed on the basis of the averaged cycle. As shown on the basis of the exemplary results (Figure 3), the spread of the maximum pressure value is not only a function of its maximum value, but also the angle of its occurrence. Obviously, the operation of a piston engine cannot be considered without taking into account its exhaust emissions.

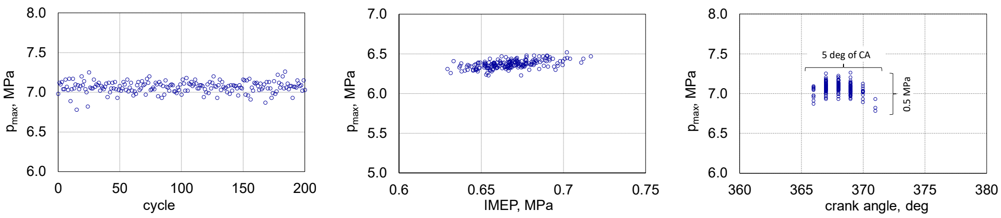


Figure 3. Example of uniqueness of p_{max} (fuel BB80).

3.1. Characteristics of the Combustion Process

Indicated results of an internal combustion engine are the best source of information about its operation. The pressure courses presented in Figure 4 are the average traces obtained for the set of 200 consecutive engine operation cycles.

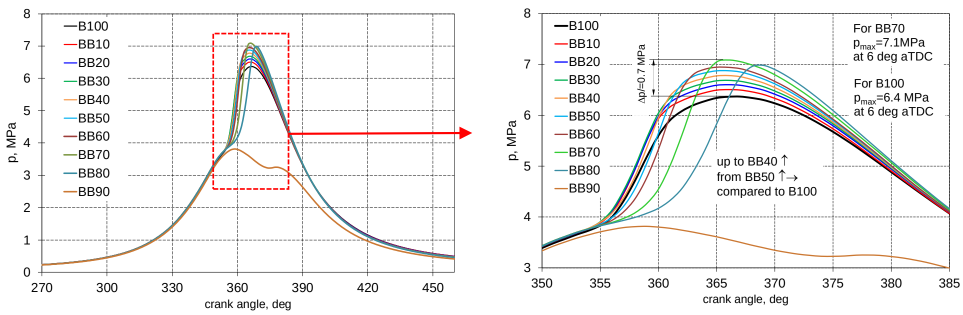
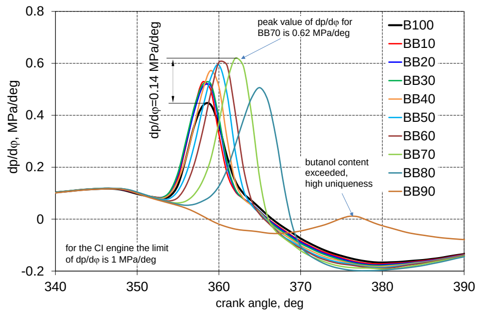


Figure 4. Mean pressure trace for analyzed cases.

Based on the analysis of pressure courses, it can be concluded that up to 40% of 1-butanol energetic fraction there was only increase in peak pressure. For fuel BB40 it was noticed that peak of in-cylinder pressure was equal to 6.88 MPa at 6 degrees aTDC. It was higher value of 0.51 bar compared to reference fuel (6.37 MPa at 6 degrees aTDC). Starting from 50% share of 1-butanol, up to 70% the increase in the value of peak pressure and its occurrence after TDC was noticed. The ignition delay is already clearly visible. In case of 80% of 1-butanol fraction in blend was observed a significant slowdown in combustion, the peak pressure value is a bit lower compared to BB70. For the last analyzed share of 1-butanol, the combustion process took place only in some cycles. This is an unacceptable share of 1-butanol for the engine.

Figure 5 presents the results of pressure increase rate and heat release rate for the analyzed range of 1-butanol shares in blend with RME. The rate of pressure increase in the engine cylinder gives information about the so-called the hardness of engine work. For compression ignition engines, the limit is 1 MPa/degree [21]. For all shares of 1-butanol, the maximum values of dp/dφ were higher in relation to the engine powered by RME (excluding BB90 blend). The highest value of dp/dφ was obtained for BB70 and was equal to 0.62 MPa/deg. It was higher value by 0.14 MPa/degree compared to reference case. Analyzing the heat release rate, it was found that the highest dQ/dφ value also occurs for BB70 and it was equal to 71.7 J/degree, which was higher by 25 J/degree compared to reference case. The integrated courses of the heat release rate allow to determine the course of heat release.



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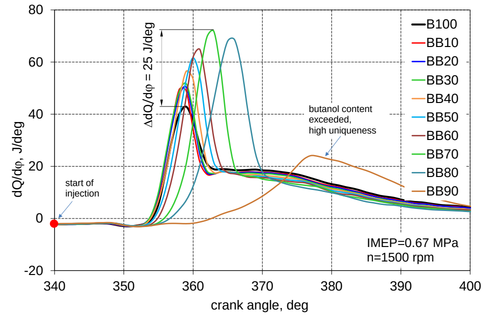
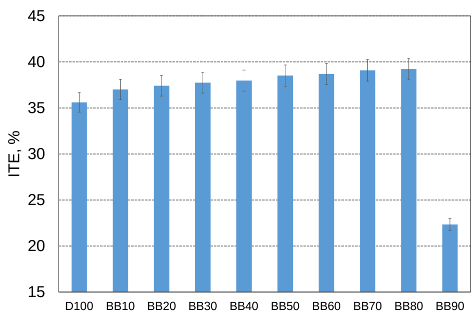


(**b**)

Figure 5. Pressure increase rate (**a**) and heat release rate (**b**).

As shown by the HRR curves in Figure 5, the combustion process takes place in two stages. In the first stage, the fuel that has evaporated during the ignition delay is combusted, it is premixed combustion. The second stage, visible in the HRR waveform, is the diffusion combustion phase. It can be stated here that the share of the premixed phase is larger in relation to the diffusion combustion phase. A similar phenomenon was observed by Zhao et al. [22]. Yesilyurt et al. [23] also obtained an increase in the maximum value of $dQ/d\varphi$ with the proportion of butanol in the mixture. The heat release curves are normalized, and they are used to determine characteristic combustion stages.

The engine was tested under constant load. The engine had to be supplied with the amount of fuel that would provide the same load. Due to the fact that for each case, the new fuel (a mixture of RME—1-butanol) had a different calorific value, instead of the specific fuel consumption, the specific consumption of energy contained in the fuel dose was used for the assessment. It is visible (Figure 2) that with the increase in 1-butanol fraction in blend the energy demand of the engine decreases, up to BB80. For the last acceptable proportion of 1-butanol in blend (BB80), obtained a decrease in SEC by almost 9% compared to the reference case (Figure 6b). That behavior can be explained by better atomization end evaporation of the injected fuel blends that have a lower viscosity with increase in 1-butanol share in the mixture compared to RME. For an engine powered by RME it was equal to 10.11 MJ/kWh and for BB80 it was 19.18 MJ/kWh. This was reflected in the thermal efficiency of the engine. With the increase in 1-butanol fraction (up to 80%) the thermal efficiency of the engine increased (Figure 6a). For engine powered by BB80 it reaches value of 39.2% and it was higher of 3.6% compared to engine powered by reference fuel. The thermal efficiency of the engine (ITE) is an important factor determining the degree of conversion of the energy contained in the fuel to the operation of the engine.



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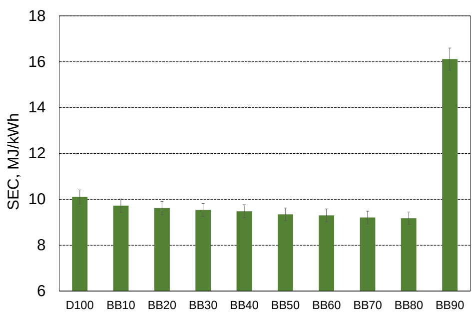


(**b**)

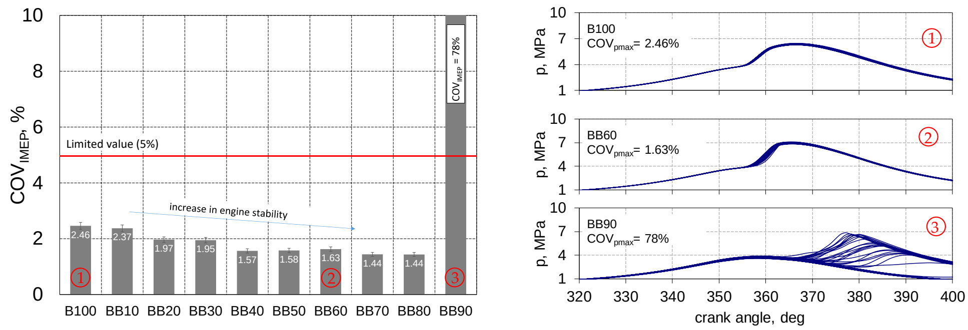
Figure 6. Indicated thermal efficiency (ITE) (**a**) and specific energy consumption (SEC) (**b**).

Summarizing this part of the analysis, it can be concluded that the proportion of 1-butanol in the mixture with RME is favorable for the engine operation. The combustion of these mixtures is gentle and predictable and gives good results: an increase in engine efficiency. Xiao et al. [15] also reported an increase in efficiency with an increase in the proportion of butanol in the mixture. Additionally, the engine allowed to combust up to 80% of the energy content of 1-butanol in the mixture. This enables flexible dosing of both fuels.

3.2. Stability of Combustion Process

Stable operation of a piston engine is a very important aspect of its characteristics. A reciprocating engine, due to its specificity, will always be characterized by some instability. It is important that this instability is within acceptable limits. The most common criterion for assessing the instability of engine operation is the uniqueness of the indicated mean effective pressure in subsequent cycles, determined by the COV_{IMEP} index. According to the literature sources, the COV_{IMEP} value should not exceed 5% for engines driving electric generators [21]. The uniqueness of the engine's operation also affects the durability of the mechanical components of the engine, generating, e.g., torsional vibrations of the crankshaft. On the other hand, engine control requires knowledge of the nature of the ignition delay variation and the combustion duration, or the angle after TDC of 50% heat release. Knowledge of these parameters gives the opportunity to optimize the engine cycle, ensuring maximum efficiency.

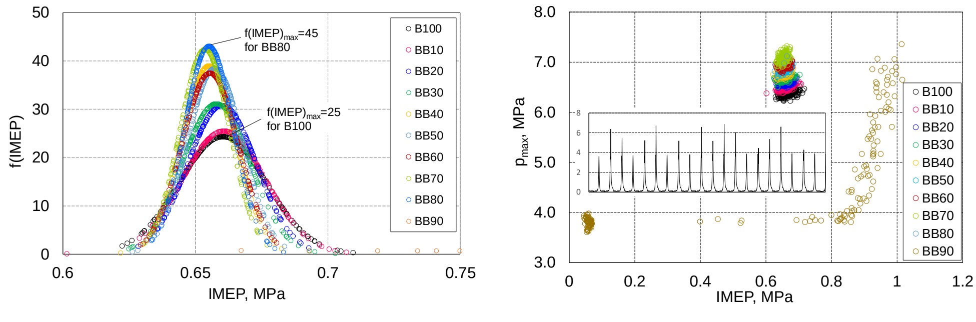
As mentioned, the basic parameter is COV_{IMEP}, the obtained data for the studied case are included in Figure 7. With the increase in the share of 1-butanol in the mixture with RME, the stability of the engine's operation increased up to 80% of energetic share 1-butanol. For the reference fuel, the IMEP uniqueness was 2.46% and for 80% of 1-butanol it was equal 1.44%. For three selected shares of 1-butanol, pressure courses from 200 consecutive engine operation cycles were presented. This shows a high repeatability of cycles for large shares of butanol.



(**a**) (**b**)

Figure 7. COV of IMEP (**a**) and cycle-by-cycle variation (**b**).

Figure 8 presents the data of probability density of IMEP and the relationship between IMEP and peak pressure. For the quality of engine operation, it would be ideal that the IMEP spread shown in Figure 8a was as small as possible. From the IMEP distributions in Figure 8a it can be seen that the mean IMEP values were slightly different. For an engine powered by RME the value of IMEP was equal 0.66 MPa but for BB80 it was 0.65 MPa. The presented distributions show that the share of 1-butanol is conducive to increasing the stability of the engine operation. For BB80 there was an almost a two times increase in the value of the function f (IMEP) in relation to B100. The analysis of the data presented in Figure 8b, shows that up to 80% of 1-butanol the spread of the peak pressure values in relation to IMEP is in a very narrow range. For the BB90, it can be seen that there is large number of cycles where combustion has not occurred or was initiated too late. There is also a large group of cycles in which, due to the large ignition delay time, cycles with a high value of IMEP were created. This could be due to, inter alia, the additional dose of fuel remaining from the previous work cycle.



(**a**) (**b**)

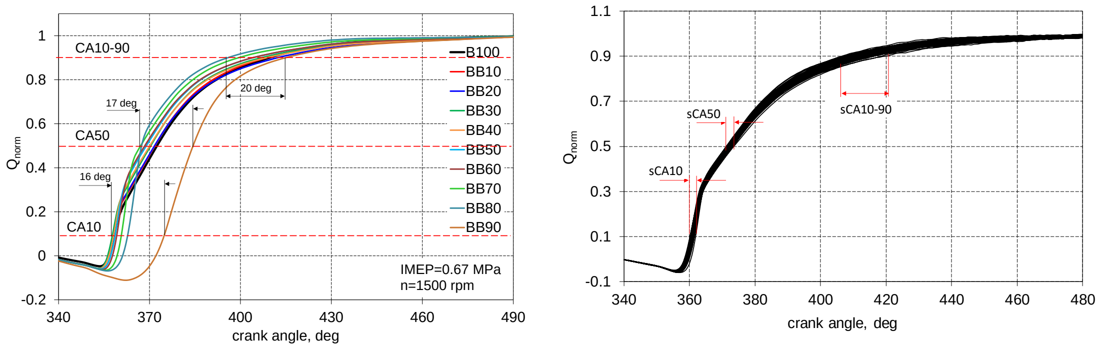
Figure 8. Probability density of indicated mean effective pressure (**a**) and relationship between IMEP and p_{max} (**b**).

For parametric evaluation of the combustion process in a reciprocating engine, three characteristic stages are taken into account. Their values are a source of valuable information for the engine control system. These characteristic stages of combustion are: ignition delay time, angle after TDC of 50% of heat release and conventional end of combustion. The time of ignition delay, characteristic for a given engine and the fuel consumed in it, should be taken into account when determining the angle of the beginning of fuel injection into the combustion chamber of the engine. Determining the angle of 50% heat release gives information about the efficiency of the engine. According to the literature, in an engine in which 50% of heat is released about 8–10 degrees aTDC, such an engine achieves its maximum efficiency. Determining the end of combustion gives information about the duration of combustion. Too long duration of combustion will contribute to a decrease in the efficiency of the engine, because the losses of heat exchange will increase. All these parameters will also indirectly influence the emission of toxic exhaust components.

The ignition delay time is defined as the period from the start of fuel injection until 10% of heat is released (CA10). The value of the ignition delay is influenced by both the chemical properties of the fuel and physical processes. Chemical factors affecting the ignition delay depend on the quality of the fuel, the C/H ratio and the ratio of oxygen in the molecular structure. Another factor depending on the type of fuel are the heat of vaporization, the ignition temperature, the laminar flame speed or the calorific value. Physical factors affecting the ignition delay are: type of fuel injection and thus fuel atomization as well as thermal-flow processes taking place in the combustion chamber of the engine.

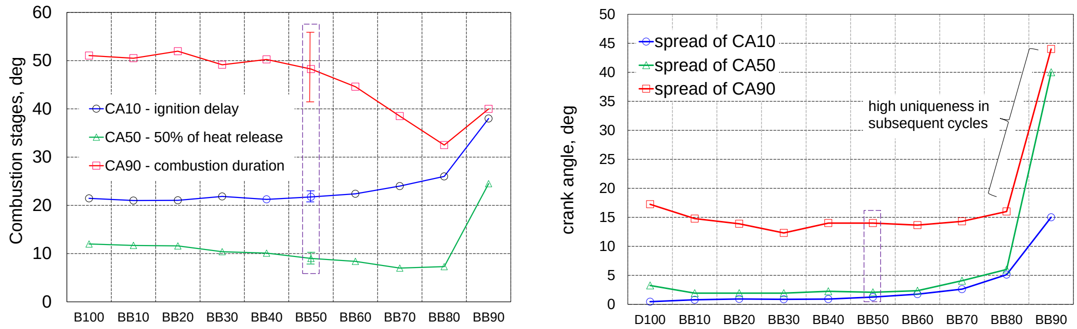
These processes mainly affect the physical ignition delay period. In the combustion chamber of the engine, both physical and chemical delay occur simultaneously.

Figure 9 presents the normalized heat release and the conventional combustion stages determined on their basis. These curves are the result of averaging over 200 consecutive engine work cycles. Due to the nature of the engine's operation, each of the characteristic combustion stages occurs individually for each cycle. The single value represents only the general nature of the changes. For a complete view of the variability of these values, the dispersion for the entire analyzed data set should be taken into account [24–26]. This is especially important when determining the conventional end of combustion, where this discharge can be very large. Figure 10a shows the combustion stages and in Figure 10b spread of these stages.



(**a**) (**b**)

Figure 9. Normalized heat release and combustion stages (**a**) and the spread of the characteristic stages of combustion (**b**).



(**a**) (**b**)

Figure 10. Combustion stages (**a**) and spread of combustion stages (**b**).

The first combustion stage is the ignition delay (ID). For the analyzed fuel blends, it was found that up to 50% of 1-butanol of energetic fraction the ID is at the same level, it was equal to 21.5 degrees. After exceeding this share of 1-butanol, a certain increase in this value is already visible. For the BB80, there was an increase in ID by 5 degrees compared to the reference fuel.

Mixtures of biodiesel and 1-butanol are characterized by higher ignition delay due to its lower cetane number and value of heat of vaporization, this means that most of the fuel can evaporate before ignition. Considering the duration of combustion (CD), it was found that up to 40% of 1-butanol fraction this parameter is stable equal to 50 degrees of CA. After exceeding the 40% share of 1-butanol, the combustion time decreased. For BB80 combustion duration was equal to 32.5 degrees and was 18 degrees shorter compared to reference fuel. This was due to the predominance of 1-butanol in the blend that combust faster than RME. The angle of release of 50% of the heat, with an increase in fraction of 1-butanol, approaching towards TDC (up to BB80). The 90% of 1-butanol is shown only as unacceptable to the engine. The increase in the ignition delay time contributes to the delay of combustion and thus the combustion process may be extended and an increase in heat loss to the engine cylinder walls can be expected. This can consequently lead to a reduction in the thermal efficiency of the engine and an increase in BSFC [27]. The dispersion of these combustion stages should be also analyzed. In case of ID spread that up to 50% of 1-butanol fraction the repeatability of the combustion starts very stable and within the limits 1.5 degrees. For larger shares of alternative fuel, the uniqueness of the distribution reaches the scope of 5 degrees for BB80. Therefore, for BB80 fuel the ignition delay can be defined as a value of 26 ± 2.5 degrees. The larges spread occurs when determining the conventional end of combustion (CA90). This stage is determined based on the already flattened part of the normalized heat release curve. As can be seen from Figure 9 and data in Figure 10 the conventional end of combustion is determined with great uncertainty.

For example, for BB80, the combustion duration is set equal to 33 degrees but the spread of this parameter is ±8 degrees of CA.

3.3. Emission Characteristics

The quality of the piston engine operation should be assessed with reference to its exhaust emissions. The research measures the emission of carbon monoxide (CO), carbon dioxide (CO_2), nitrogen oxides (NO_x), unburned hydrocarbons (THC) and soot. The other important factors affecting in-cylinder combustion and emission is the oxygen concentration in the combustion chamber. Oxygen concentration, whether overly lean or rich causes such an increase. Lower viscosity of blends increases spray penetration.

In Figure 11 are presented results of exhaust gases analysis of engine powered by RME—1-butanol blends. In a compression-ignition engine, carbon monoxide (CO) emission depends on local insufficient air, problems with the formation of a combustible mixture and too short a time for CO to CO_2 oxidation. Due to the fact that in a compression-ignition engine, combustion takes place with excess air, the emission of carbon monoxide (CO) is lower than in a spark-ignition engine. With the increase in 1-butanol fraction in blend CO emission is lower in comparison to the diesel fuel combustion. This is because there is no oxygen in the molecular structure of diesel fuel. The oxygen content in the fuel structure intensifies combustion. Temperature and the presence of oxygen molecules have a key influence on the oxidation of CO to CO_2. In this case, the oxygen contained in 1-butanol contributes to the reduction of CO emissions. In case of BB80 the decrease in CO emission reduced to 0.178% and it was over 3.7 times lower compared to reference fuel. In the oxidation of carbon particles, takes part not only oxygen from the air but also oxygen from alcohol particles. High CO_2 content in the exhaust gas gives information about the quality of the combustion process. Due to the higher engine efficiency CO_2 also decreased because less fuel is burned. CO_2 is one of the most important greenhouse gasses produced by internal combustion engines that causes global warming.

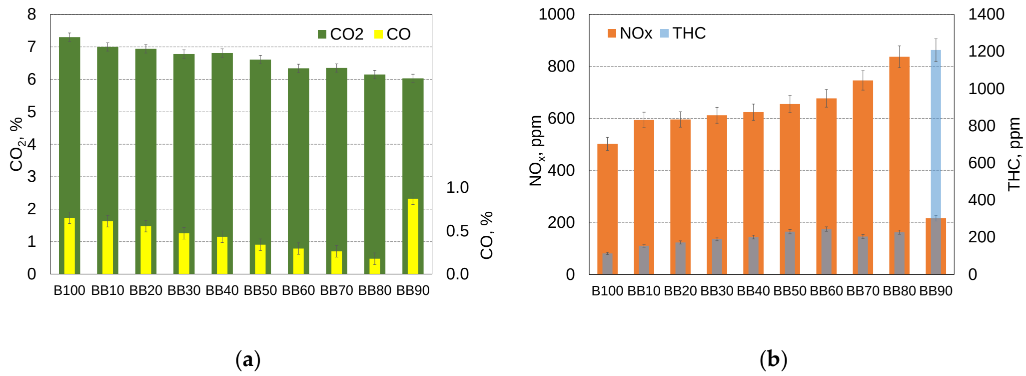


Figure 11. Engine exhaust emissions CO and CO_2 (**a**) and NO_x and THC (**b**).

THC emission is mainly due to trapping the fuel in the crevice volume of a combustion chamber, lowering the combustion temperature, locally overreach or over lean air-fuel mixtures in the combustion chamber, quenching the flame on the wall of the combustion chamber or incomplete fuel evaporation. The main factors contributing to the production of THC are oxygen deficiency, locally rich mixtures and too low temperature. The reason for too low temperature in the engine combustion chamber may be too much heat loss to the engine cooling system. Near the cold cylinder walls may occur quenching the flame. For the analyzed cases, the highest THC emission was measured for the combustion of the BB60 blend and it was equal to 244 ppm. It was 100% more compared to the reference fuel. Up to 60% of 1-Butanol the THC emission increased, then, up to the BB80, emissions were practically the same. For the BB90 it was very high, but this is due to the disappearance of the combustion process. Due to the high LHE value of alcohol, the in-cylinder temperature is lowered, and consequently, it leads to incomplete combustion and an increase in THC emissions.

The variation of nitric oxide emission with increase in 1-butanol fraction is depicted in Figure 11. Due to the fact that diesel engines operate with excess air, they are susceptible to increased emissions of nitrogen oxides. The formation of NO_x depends on oxygen concentration, temperature in the combustion chamber and residence time. An oxygenated fuel such as a RME and 1-butanol improves fuel oxidation process in the reaction zone which causes higher local temperatures. It can be observed from Figure 11 that the NO_x emission of RME-1-butanol blend combustion, up to BB80, is higher compared to that of the reference fuel. With the increase in the share of 1-butanol, NO_x emissions increased successively. The highest value was obtained for BB80 and it was 837ppm. It was 67% more compared to the combustion of the reference fuel. Figure 12 are presents the results of soot emission estimation. The smoke is formed due to incomplete combustion. As the 1-butanol rate increases in the blend, soot emission decreased.

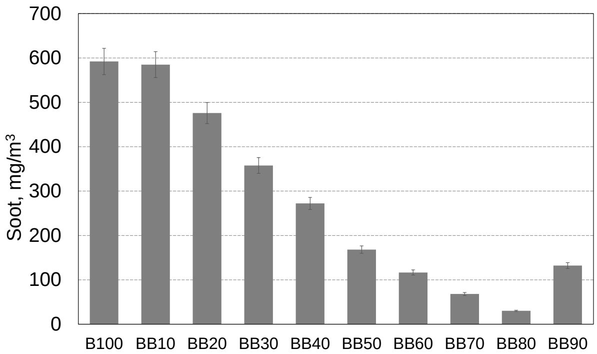


Figure 12. Soot emission.

Reduction in soot emissions is principally based on the oxygen content of fuels. Butanol is characterized by a 2 times higher content of oxygen in the structure [28]. The increase in the fraction of 1-butanol in the blend resulted in higher maximum pressure values, which caused an increase in temperature in the engine cylinder. Combustion of fuels containing oxygen in the molecular structure, leads to a reduction in the formation of soot. Along with the increase in the share of 1-butanol in the fuel mixture, a significant decrease in soot emissions was found. For the 50% energetic share of 1-butanol, the soot emission was over 3.5 times lower. Improvement of combustion caused a decrease in smoke density. The incomplete combustion is a source of soot emissions. The formation of soot in the combustion chamber of the engine begins by pyrolysis in fuel-rich zones of fuel spray during the diffusion combustion phase.

4. Conclusions

The paper presents the results of the evaluation of the combustion process and exhaust emissions of an engine powered by a mixture of RME and 1-butanol. Much attention was paid to assessing the combustion stability by determining the spreads of the characteristic combustion stages. The results were obtained for the engine working at a constant load and with the same angle of fuel injection to the combustion chamber. In terms of the assessment of the combustion process, it was found:

- Up to 40% of 1-Butanol energetic fraction there was only increase in peak pressure, for larger shares of 1-butanol, it was an increase in the value of peak pressure and was moved further from TDC;
- The highest value of $dp/d\varphi$ and $dQ/d\varphi$ was obtained for BB70 and was equal, respectively, 0.62 MPa/degree and 71.7 J/degree;

- With the increase in 1-butanol fraction in blend the energy demand of the engine decrease, up to BB80, what result in decrease in BSEC by near to 9% compared to reference case;
- With the increase in 1-butanol fraction (up to 80%) the thermal efficiency of the engine increased, for engine powered by BB80 it reaches value of 39.2% and it was higher by 3.6% compared to engine powered by reference fuel;
- With the increase in the share of 1-butanol in the mixture with RME, the stability of the engine's operation increased up to 80% of energetic share 1-butanol;
- The share of 1-butanol is conducive to increasing the stability of the engine operation, for BB80 there was an almost double increase in the value of the function f (IMEP) in relation to B100;
- Up to 50% of 1-butanol energetic fraction ID is at the same level equal to 21.5 degrees, exceeding this share of 1-butanol, ID increases, for the BB80, there was an increase in ID by 5 degrees compared to the reference fuel;
- Up to 40% of 1-butanol fraction CD is stable and equal to 50 degrees of CA, exceeding the 40% share of 1-Butanol, the combustion time is decreasing;
- In case of ID spread that up to 50% of 1-butanol fraction the repeatability of the combustion start was very stable and within the limits 1.5 degrees, for larger shares of alternative fuel, the uniqueness of the distribution reaches the scope of 5 degrees for BB80;
- The end of combustion is determined with great uncertainty, for BB80, the combustion duration is set equal to 33 degrees but the spread of this parameter is 8 degrees of CA.
- In terms of the assessment of the emission, it was found:
- In case of BB80 the decrease in CO emission reduced to 0.178% and it was over 3.7-times lower compared to reference fuel;
- The highest THC emission was noticed for the combustion of BB60 blend and it was equal to 244 ppm. It was 100% more compared to the reference fuel;
- The highest value of NO_x emission was obtained for BB80 and it was 837ppm, it was 67% more compared to the combustion of the reference fuel;
- The participation of 1-Butanol in the co-combustion process with RME has a positive effect on the soot emission, already for the 50% energetic share of 1-butanol, the soot emission was over 3.5 times reduced.

Summarizing these studies, it can be stated that 1-butanol is a fuel friendly to a compression ignition engine. It is true that the engine cannot be powered by itself, but mixed with RME, it can be burned up to its 80% energy share. An increase in the proportion of 1-butanol causes an increase in engine efficiency, improves work stability (COV_{IMEP}) and significantly reduces soot emissions.

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